PERFORMANCE EVALUATION OF A CENTRIFUGAL PUMP SYSTEM UNDER VARIABLE SPEED OPERATION

C. Kühn* and PJ. Randewijk**

**Electrical Engineer at Langeberg Municipality, Ashton, South Africa Email: curren.kuhn@live.com*

***Department of Electrical and Electronic Engineering, Stellenbosch University Email: pjrandew@sun.ac.za*

Abstract: The aim of this research project investigates the performance of a centrifugal pump system under variable speed operation. Operating a pump under variable speed operation brings the affinity laws into play. These equations were firstly investigated before it was applied to the overall investigation of pump performance. Distinction is made between three types of systems, a system with high static head (lift system), the second system is a high loss system with no- or little static head and the third is a mixed type system. The “correct” affinity laws were then used to apply to a system with a minimum pressure requirement. The theory proved accurate and was confirmed by practical findings. It has been found that both the pump and the VSD experience a slight drop in efficiency for small changes in speed. A maximum of 5% drop occurred in the efficiency of the pump for a 30% reduction in speed. Further reductions in speed resulted in a more rapid drop in efficiency.

Keywords: Variable speed drive (VSD), pump efficiency, system curve, static head, power consumption.

1. INTRODUCTION

Many aspects of irrigation pumping systems consist of compromises or trade offs between first costs of the system and the running costs. Purchasing cheaper systems without considering the higher running cost is a tendency among users resulting in a widespread use of, inefficient and non cost-effective systems. In terms of running costs a significant portion of the operational cost in water distribution systems can be related to pumping [1]. Therefore, variable speed pumping has been a recent consideration with the aim of varying the duty point of the pump to match delivery rate to demand. Depending on the system characteristics, this approach can lead to considerable savings in operational costs. In particular cost reductions, where advantage can be taken of the demand variability leading to a significant decrease in energy consumption.

Pumping systems are usually designed to meet maximum design discharge which might occur just for a limited time [2]. In on-demand systems such communal water supply, there is a variation in flow and pressure as a result of seasonal changes in water requirements and human behaviour. The energy consumption of the system depends on the systems flow rate, operating pressure and the period of time the system operates. Savings can be realized by reducing theses variables, either by changing the system’s characteristics or the pump’s characteristics.

2. PIPELINE SYSTEMS

The relationship between the flow in a pipeline and the head loss produced is described by the system curve of the pipeline. The essential elements to include in a system design, is the static head and the friction head. Adding the static head to the friction head losses as the flow increases gives the total head (H) and is essentially a parabola with its origin at the value of the static head.

$$H = K_1 Q^{1.85} + K_2 Q^2 + H_s \tag{1}$$

with:

K_1 = some constant indicating the system’s resistance

Q = delivery rate in m^3/h

K_2 = some constant indicating the losses due various pipe fittings

H_s = static head in m

Distinction is made between three types of systems, a system with high static head (lift system), the second system is a high loss system with no- or little static head and the third is a mixed type system [3]. These are illustrated in Figure 1. The head-capacity curve of the pipeline to which the pump is installed is called a system curve and determines the performance of the pump. In order for the pump to achieve flow the pump must overcome the opposing head represented by the system curve. The opposing head consists of static head, frictional head losses in the pipeline and the type of liquid being pumped. Static head is the difference in height of the supply and the discharge which is independent of the flow rate. Friction head refers to friction loss on the liquid being moved in the pipeline and the various pipe fittings.

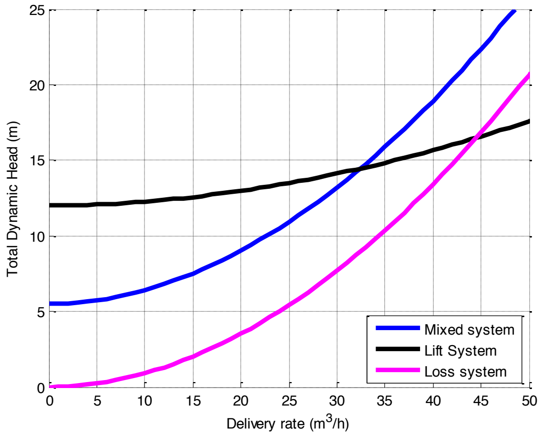

Figure 1: The three types of systems identified

The system was initially designed for a static head of 12 meters, but after applying the affinity laws for speed relating to pressure and flow rate, it was noticed that the operating region for variable speed pumping was greatly reduced. Not allowing the speed to be varied less than about 70% of the nominal speed. For the purpose of this study it was necessary to vary the speed to at least 50% of the nominal speed. To achieve this, the static head was lowered to 5.5 meters and allowed the speed reduction to reach 55%.

3. SYSTEM DUTY POINT

Pump performance curves are usually determined at a constant impeller rotational speed by varying the flow (Q) by means of throttling. These characteristic curves are plotted as head (H) against flow where the head or pump pressure is measured in meters (m) and the flow is measured in cubic meters per hour (m^3/h). The head-capacity curve of the pump and the system curve are plotted on one graph and the operating point of the pump is found at the intersection of these plots, Figure 2.

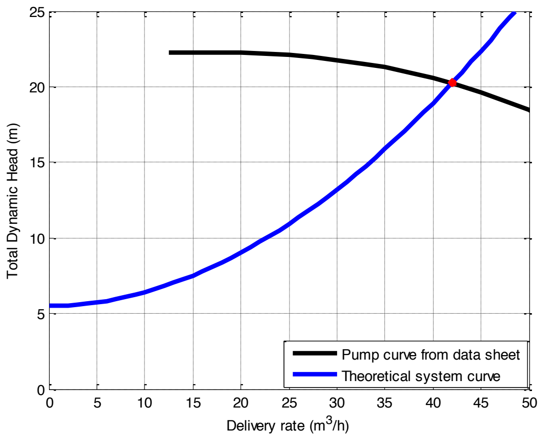

Figure 2: Pump operating point in a given system

The intersection is the head and capacity at which the pump would operate at a certain shaft speed in a given piping system. Distinction is made between three kinds of power, supplied power from an external source to the motor and controller denoted P_{in}, shaft power from the motor to the shaft denoted P_{shaft}, and hydraulic power imparted from the impeller to the fluid denoted P_{hyd}. The hydraulic power can be calculated at the pump's operating point by,

$$P_{hyd} = \frac{Hg\rho Q}{3600} \qquad (2)$$

with:

H = head in m

Q = delivery rate in m^3/h

ρ = fluid density (998.2 kg/m^3 at 20 ^oC for water)

g = gravitational constant 9.81 m/s^2

The power required to drive the pump shaft at the operating point can be expressed as

$$P_{shaft} = \frac{P_{hyd}}{\eta} \qquad (3)$$

with η being the efficiency of the pump at the duty point.

3.1 Changing the System Duty Point

When varying the speed of a pump the basic system characterisation stays unaffected with only the pump characteristics changing. Changing the impeller rotational speed is accomplished by changing the drive motor's shaft speed, usually by means of a Variable Speed Drive (VSD) and brings the affinity laws into play. This is a set of equations that can be applied to estimate the performance of the pump under variable speed operation. During variable speed operation, both system head and flow rate reduces for a lower drive speed which results into lower power consumption. These laws states as follows:

- Flow is proportional to speed ($Q \propto N$),
- Head is proportional to the square of the speed ($H \propto N^2$),
- Required shaft power is proportional to the cube of the speed ($P \propto N^3$).

It has become somewhat of a "fashion" to install VSD's into pumping systems, due to the third affinity law;

power is proportional to the cube of the speed. A common first impression is that centrifugal pumps are more efficient at lower speeds, which is not necessarily true. These laws do not make any statement of the efficiency of the pump. Understanding these laws and why they work is key in applying them correctly.

4. AFFINITY LAWS

In order to use the "correct" affinity laws to estimate the power using the flow rate, some adjustments needed to be made to the power equation for systems with static head [4].

$$\frac{P_2}{P_1} = \left[\left(1 - \sqrt{\frac{H_{\min}}{H_N}} \right) \frac{Q_2}{Q_1} + \sqrt{\frac{H_{\min}}{H_N}} \right]^3 \tag{4}$$

Where:

P = required power in W

H_{min} = minimum system head (static head) in m

H_N = duty point total head in m

Equation (4) incorporates the minimum pressure requirement in a given piping system for estimating the power requirements at a given duty point. It can be seen that as H_{min} decreases towards 0, equation (4) becomes that of the original affinity laws stated earlier.

5. SYSTEM MODEL

Discussing the efficiency of centrifugal pumps revolves around four efficiencies [1],

- Hydraulic efficiency,
- Mechanical efficiency,
- Drive efficiency,
- The overall system efficiency which is the product of the above.

Mechanical efficiency is a measure of the losses between the drive shaft and shaft input at the impeller and relative to other losses mechanical losses are small and are usually ignored [5]. Drive efficiency refers to the effectiveness of the pump driver which includes the motor and the VSD. Thus, the total drive efficiency will be the product of the motor efficiency and the VSD's efficiency, Figure 3.

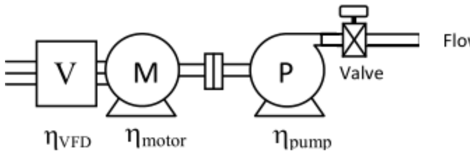

Figure 3: Pump system block diagram [1]

6. THEORETICAL ESTIMATES OF COMPONENT EFFICIENCY

6.1 VSD

The efficiency of the VSD is based on the research done by the Irrigation Training and Research Center [6]. The study concluded an efficiency of 95% for a VSD manufactured by Danfoss.

6.2 Three- phase induction motor

Determining the motor's efficiency was done using the method of constant volts-per-hertz (V/f). The input voltage to the motor is adjusted proportionally with the input frequency, keeping the flux constant in the machine. The model of Figure 4 was used with the accompanied equations. The motor parameters were obtained using the Automatic Motor Adaption (AMA) feature of the Danfoss VLT® FC302 Automation Drive. The parameters are shown in Table 1.

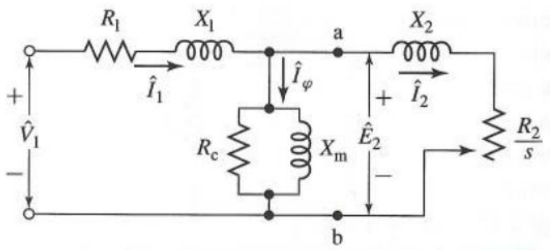

Figure 4: Per phase induction motor equivalent cicuit [7]

Table 1: Induction motor parameters

R_1 [Ω]	X_1 [Ω]	R_2 [Ω]	X_2 [Ω]	R_c [Ω]	X_m [Ω]
2.05	3.04	1.85	3.04	1466	60

Using Table 1's values the model was simulated to obtain the efficiency of the induction motor under variable speed operation.

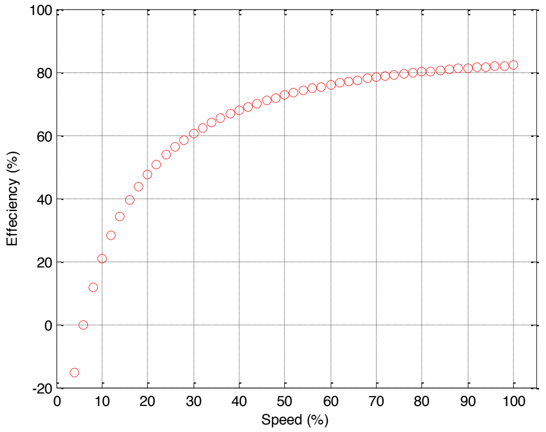

Figure 5: Induction motor efficiency for constant V/f control

It can be seen that the motor efficiency is about 82.25% at nominal speed and decreases slightly towards a 74.8% efficiency at 55% speed. The motor nameplate rating indicates an efficiency of 83% at full speed which compares well with the result obtained. It is noted that the drop in motor efficiency becomes more rapid for speeds lower than 60%.

6.3 Pump

Using the affinity laws, Figure 6 was obtained. Table **2** contains the data obtained from Figure 6. The hydraulic power is calculated at each duty point using equation (2) and will further be used to determine the pump's efficiency.

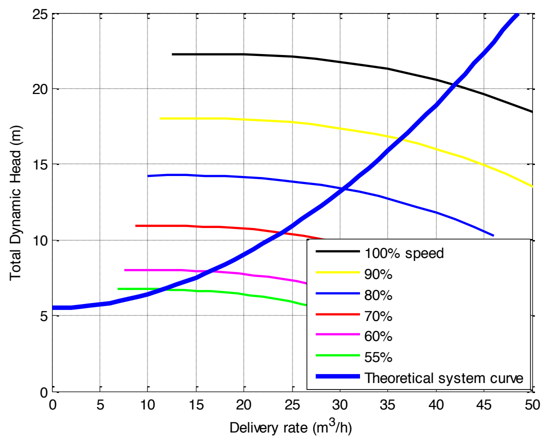

Figure 6: Estimate duty points for various speeds

Table 2: Theoretical estimates at various duty points

	Duty point		
Speed *N* (%)	Flow *Q* (m^3/h)	Total Head *H* (m)	Output Power P_{hyd} (W)
100	42	20.26	2314.58
90	36	16.67	1632.38
80	30	13.41	1094.29
70	24	10.5	685.46
60	16	7.88	342.95
55	11	6.61	197.78

A complete table of the theoretical system estimates is given in Table 3.

Table 3: Complete system analysis

		Efficiency		Power (W)	
Speed *N* (%)	VSD	Motor	Pump	Shaft power	Total input
100	95	82	70	3307	4232
90	95	81	61	2673	3462
80	95	80	53	2083	2740
70	95	78	45	1540	2069
60	95	76	38	914	1266
55	95	75	35	562	791

The above table indicates a change in the pump's efficiency under variable speed operation, with a maximum drop of 9% for each 10% decrease in speed. Theoretical values indicate a drop in pump efficiency but needs to be confirmed by practical data obtained from testing the designed system.

7. PRACTICAL FINDINGS

7.1 Test Setup

Tests were conducted in a sequential manner. Starting at nominal operating speed the speed was reduced in steps of 10% of the nominal speed down to 55% speed. The total input power to the system, e.g. to the drive, was measured using a three-phase power measuring device. The drive provided a function to view the input power to the motor making it possible to determine the efficiency of the drive itself. Further readings were the flow rate, digitally displayed on the flow rate measuring device, and the pressure displayed in volts on a voltmeter. After the test readings were recorded at the duty points, the system was throttled to determine the pump curve at that specific speed. Five throttling stages were deemed sufficient for determining the pump curve. This process was repeated at all reduced speeds. The constructed system and measurement setup are shown in Figure 7 and Figure 8 respectively. The vertical elevation (static head) between pump suction and the discharge in the top tank is 5.5 meters.

Figure 7: Constructed pump system

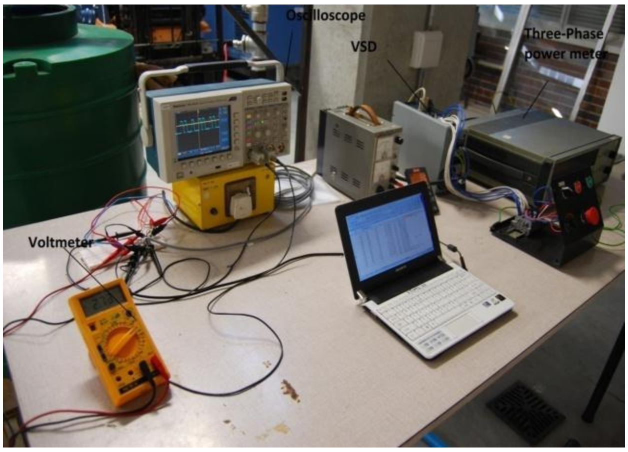

Figure 8: Test setup

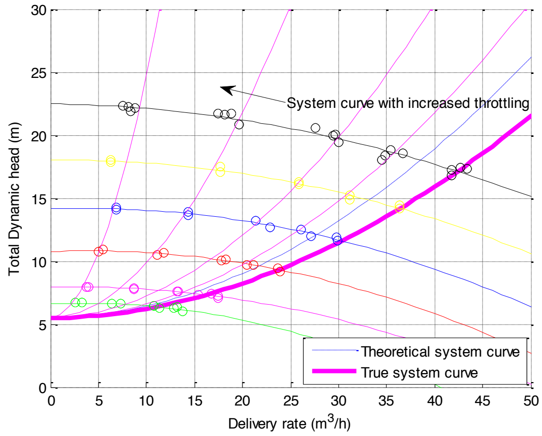

Figure 9: True measured pump curves

Also using Figure 9, a comparison of the average energy consumption between the throttled system and variable speed operation is shown in Table 4. The throttling occurs at 100% speed. The valve is first fully open and then it gets partially closed for the five different throttling stages. It can be seen that the savings on energy consumption increases for lower speed operation compared to the increased throttling. From 17.9% to as much as 64.45% energy savings occurred at reduced speeds for variable speed operation.

Table 4: Comparison of energy consumption between a throttled system and variable speed operation

Variable Speed			Throttled System		Savings (%)
Speed N (%)	Input Power (W)	Flow Q (m^3/h)	Input Power (W)	Flow (m^3/h)	
55	743	11.82	2090	8.34	64.45
60	960	17.54	-	-	-
70	1485	23.64	3440	19.62	56.83
80	2213	29.33	3630	30.06	39.04
90	3128	35.87	3810	34.56	17.90
100	4442	43.08	4442	43.08	0.00

7.2 Results

In Figure 9 the true pump curves are plotted with the theoretical system curve. The system was designed for the pump to operate close to its best efficiency point. It can be seen that true resistance of the system is a bit lower than the theoretical. This also explains the lower head measured at nominal speed.

7.3 Efficiency

The efficiency of both the VSD and the pump are plotted in Figure 10. It is evident that both the component's efficiencies decrease with the speed being varied. Working with the extreme values it can be seen that an efficiency drop of no more than 7% occurred for a total of three consecutive 10% reductions in speed. Below 70% speed the drop in efficiency becomes more rapid with a maximum drop of 19% occurring for the speed being reduced a further 15%. For the VSD a drop of 6% occurred for the entire speed range.

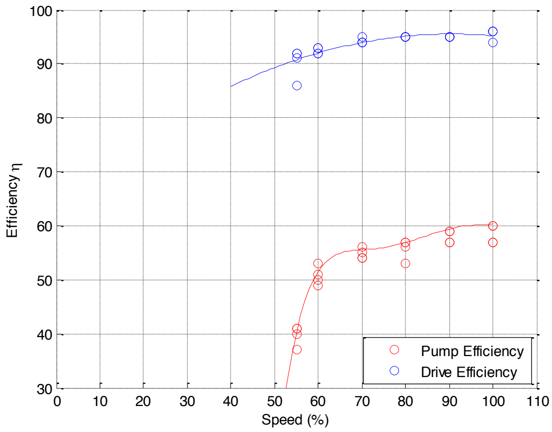

Figure 10: Efficiency plot of the VSD and pump

The average values are plotted in Figure 11 with constant efficiency lines. It is noted at nominal conditions, that the pump efficiency is about 58% instead of 70% as indicated by the pump data sheet. This can be ascribed to the drop in head measured. From Figure 11 it can be seen that as the speed is reduced the pump duty point moves from 58% constant efficiency line towards the 40% constant efficiency line. The drop in efficiency can be seen to be more rapid for speeds lower than 70%.

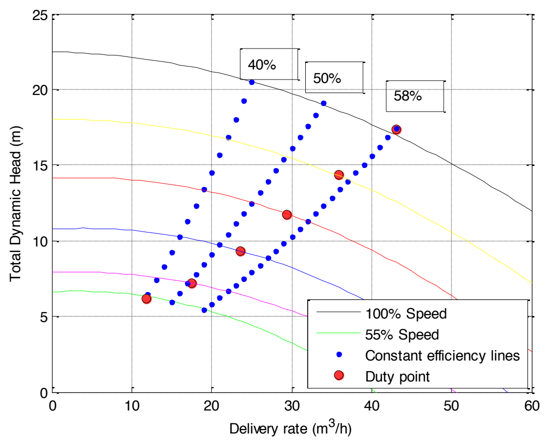

Figure 11: Pump operation with constant efficiency lines

8. CONCLUSION

Having considered the theoretical calculations and the practical findings on the pump and VSD efficiency, it can be concluded that a definite change in efficiency occurs under variable speed operation. The change in pump efficiency is mainly because of the system duty point movement, with VSD losses and motor duty point movement also having an effect. Practical findings in pointed out that over small variations in speed, a slight reduction in the pump's efficiency occurred. An approximate 5% drop in pump efficiency takes place for a 30% decrease in speed. However, at a speed lower than 70% a more rapid drop in efficiency occurred.
At 70% speed a considerable amount of power can be saved. This being the case, variable speed pumping can be highly beneficial for on demand systems where demand varies with time and behaviour.

The system's efficiency increases as the speed is reduced accompanied by a reduction in flow. For systems with low- or no static head, benefits from VSD's thus seem promising. But for systems with high static head the benefits of VSD's is greatly limited. A proper analysis on the system should be done to determine the viability of a VSD controlled pumping system. Also, if energy efficiency is a major concern, then a throttling system would be out of the question.

The authors would like to thank Eskom for the financial support for this research via their TESP program.

9. REFERENCES

[1] A. Marchi, A. Simpson and N. Ertugrulm, "Assessing variable speed pump efficiency in water distribution sytems", Drink. Water Eng. Sci., Vol. 5, pp.15-21, 2012

[2] F. Barutcu, N. Lamaddalen and U. Fratino, "Energy Saving For A Pumping Station Serving An On-Demand Irrigation System", Irrigation Science, Vol. 30, Issue 2, pp.157-166, 2012

[3] Wallbom-Carlson, "Energy Comparison, VFD vs. On-Off Controlled Pumping Stations", Scientific Impeller, pp. 29-32, 1998

[4] Vaillencourt, R.R, "The Correct Formula for Using the Affinity Laws When There Is a Minimum Pressure Requirement", Energy Engineering, Vol. 102, No. 4, pp. 32-46, 2005

[5] W. Randall and P. Whitesides, "Basic Pump Parameters and the Affinity Laws," 2012. [Online]. Available:http://www.pdhonline.org/courses/m125/m125content.pdf [Accessed 7 September 2012].

[6] C. M. Burt, X. Piao, F. Gaudi, B. Busch and N. Taufik, "Electric Motor Efficiency under Variable Frequencies and Loads", ASCE Journal Irrig. Drain. Engr 134(2), pp. 129-136, 2008

[7] A. Fitzgerald, C. Kingsley and S. Umans, "Induction Motor Model and Analysis," in *Electric Machinery*, New York, McGraw-Hill, 2003, pp. 313-320.